International Journal of Mechanical and Production Engineering Research and Development (IJMPERD)
ISSN (P): 2249-6890; ISSN (E): 2249-8001
Vol. 8, Issue 2, Apr 2018, 121-130

OPTIMIZATION OF ELASTOMERIC PASSIVE ENGINE MOUNT USING DIRECT OPTIMIZATION METHOD

KIRAN VIJAY KUMAR[1] & NAGARAJA SHETTY[2]

[1]Research Scholar, Department of Mechanical and Manufacturing Engineering, Manipal Institute of Technology, MAHE, Manipal, Karnataka, India

[2]Assistant Professor, Department of Mechanical and Manufacturing Engineering, Manipal Institute of Technology, MAHE, Manipal, Karnataka, India

ABSTRACT

An ideal engine mount should be able to completely isolate the engine vibration which is caused due to the engine Noise, Vibration and Harshness characteristics. Vibration in an automobile is due to different parts of the automobile, one of the main sources of vibration is the engine. In engine, the vibration is mainly due to drive trains. Hence, it is very important to optimize the engine mount to reduce vibration at the source itself, to increase the comfort of the passenger. In this paper, standard passive engines mount which is available in market is used for optimization which is fed into the Ansys for shape optimization by fixing the surface to be retained and necessary boundary condition is applied to obtain an optimized shape, both of the models i.e. standard model and optimized model is subjected to dynamic analysis to obtain the stiffness. The stiffness obtained in optimized model was less when compared with that of the standard model. Since the shape and stiffness of the mount are optimized, the vibration transferred from the engine to the compartment is reduced as well as the life of the mount is also improved.

KEYWORDS: Passive Engine Rubber Mount, Finite Element Method, Elastic Characteristics, FEA Modeling, Optimization & Stress Distribution

Received: Jan 18, 2018**; Accepted:** Feb 08, 2018**; Published:** Feb 26, 2018**; Paper Id.:** IJMPERDAPR2018013

Original Article

1. INTRODUCTION

An ideal engine mount should be able to isolate engine vibration. The engine vibration is caused due to the engine disturbance force in the engine speed range [1]. There are three types of vibration and sounds in the automobile structure, they are (i) vibration, (ii) Noise and (iii) squeak and rattle. In an automobile, vibration is due to different parts of automobile, one of the main sources of vibration is engine. Hence, it is very important to optimize the engine mount to reduce vibration in the source itself, to increase the comfort of the passenger [2]. Six degrees of freedom is possible in a passenger car engine through which vibratory motion of the engine takes place. (i) Bounce and yaw motion in Y direction (2 degrees of freedom), (ii) Bounce, aft and roll in horizontal plane X direction (2 degrees of freedom) and (iii) Side way shake and pitch Z direction (2 degree of freedom). All these degrees of freedom lead to vibration in engine, X is the direction along the crank shaft, Y is the direction perpendicular to the crank and along top of the engine, and Z is direction along, sideways of the engine [3].

From several decades, engineers are trying to produce a perfect automobile by perfecting each and every one of its component. The increased vibration in the automobile will cause damage to the car chassis and causes discomfort to the people riding on it, the vibration damping can be done by providing a suitable mount [4]-[6]. A

mount is used to separate engine from the chassis of automobile. Hence, mount prevents the vibration to move from the engine to the chassis (body) [5]. Majority of the research done on the engine mount are by considering the ideal and uniform load conditions. Thus, the optimal mount property obtained may or may not be applicable to engine in practical situation since in reality mount may undergo distinct excitation force and varying load [6].

In the 80's and 90's most of the research were done to perfect the engine performance parameters they didn't concentrate on the comfort in case of Noise, Vibration etc. As the technology grow people asked for comfort in their ride, so now a day's research is going on to increase the comfort of the ride by reducing the most of the noise and vibration. And also most of the research is done to reduce vibration in small or average size engines, only some research is done for the heavy and bulky diesel engines. Since the diesel engine is bulky it has more vibration than any other engine [7]. In cars and trucks, the transverse mount model is most widely used rather than rotational modeled mounts because they control the rotational motion of the engine around the crank shaft. The very high torque produced in the engine tends to rotate the engine [8].

The stiffness of the engine mount is also an important factor to be considered when we want to reduce the vibration propagation from the engine to the passenger compartment. The stiffness of the engine mount should be higher when the engine operates at a lower frequency and vice versa to withstand the impact weight and also to attain vibration isolation. Such kind of engine mount is known as an ideal engine mount [1]. Hence, in reality there should be a tradeoff between frequency and the stiffness so that the mount could absorb shock as well as to reduce vibration [1]-[9].

Hence, the engine mounts need to be constrained properly, even though the engine has some degrees of freedom to vibrate since the complete vibration isolation is not possible in engine till date. In any case the passenger should not experience the vibration in passenger compartment [1]-[2]-[4]-[6]-[7].

The main objective of the paper is to optimize the shape of the engine mount and also to optimize the stiffness. A high stiffness or high damping elastomeric passive engine mount can yield a low shake level at low frequency, but its performance at high frequency will be poor. On the other hand, low stiffness and low damping yields low noise levels, but it induces a high shake level at low frequency because of the shock excitation. So the main objective of the paper is to optimize stiffness in the direction of engine vibration.

2. METHODOLOGY

In this paper, static stress strain analysis has been carried out to determine the stiffness characteristics of the passenger car rubber engine mount. To optimize the parameter of the existing model the top and bottom portion of the engine mount has been modified. Since the top and bottom portion of the rubber take more load it has to be more rugged than that of the bottom portion and also top portion under goes more deformation when compared to the bottom part. Hence, the top portion of the existing model worn out fast hence reducing the life of the engine mount, so there is a frequent need for replacement of the engine mount. So, we optimize the design such that the top portion should be able to sustain the load.

2.1. Objective Function

The target work objective function (U) for this review is characterized as the total of the Root Mean Square (RMS) estimations of the transmitted strengths and moments from bounce, roll and pitch vibrations. By applying weighting elements for the RMS values, one can change the processed values in various frequencies to take into consideration their

centrality in a specific setting.

Whatever remains of the transmitted vibration and minute values in the two recurrence groups additionally assume a critical part with a specific end goal is to lessen the transmitted vibration in the whole recurrence run. Along these lines, the aggregate of the comparing powers is additionally taken into consideration for goal of our work. The transmitted force and moments in bounce, rotate and pitch axis at low and high frequencies are represented in four separate equations, such that:

- **Weighted Three Peak Values of Transmitted Force and Moments at Low Frequency Band are Expressed As:**

$$F_{ws1}= \alpha_{21}[\max(F_{wi\_rms};i= 1,2,...\ 10)] + \alpha_{22}[\max(M_{wxi\_rms};i= 1,2,...\ 10)]+ \alpha_{13}[\max(M_{wyi\_rms};i= 1,2,...\ 10)] \quad (1)$$

Where, *i* is the number of the selected frequencies, *i* is from 1 to 10 in low frequency band.

- **Weighted Three Maximum Values of Transmitted Force and Moments at the High Frequency Band are Given As**:

$$\left.\begin{aligned} &F_{ws2}= \alpha_{21}\ [\max\ (F_{wi\_rms};i= 11,12,...\ 20)] + \alpha_{22}\ [\max(M_{wxi\_rms}\ ;i= 11,12,...20)] \\ &+\alpha_{23}\ [\max\ (M_{wyi\_rms}\ ;i= 11,12,...\ 20)] \end{aligned}\right\} \quad (2)$$

- **Weighted Sum Value of Three Transmitted Force and Moments at Low Frequency Band, Excluding the Frequencies Selected in F_{ws2}, Is Described As:**

$$F_{wx1} = \alpha_{31}\left[\sqrt{\sum_{i=1}^{10} F_{wi\_rms}^2} + \sqrt{\sum_{i=1}^{10} M_{wxi\_rms}^2} + \sqrt{\sum_{i=1}^{10} M_{wyi\_rms}^2}\right] \quad (3)$$

- **Weighted Sum Value of Three Transmitted Force and Moments at High Frequency Band, Excluding the Frequencies Selected in F_{ws2}, Such That**:

$$F_{wx2} = \alpha_{32}\left[\sqrt{\sum_{i=11}^{20} F_{wi_{rms}}^2} + \sqrt{\sum_{i=11}^{20} M_{wxi\_rms}^2} + \sqrt{\sum_{i=11}^{20} M_{wyi\_rms}^2}\right] \quad (4)$$

- **The Objective Function Is Finally Formed By the Four Terms Described Above, Such That:**

$$U = F_{ws1} + F_{ws2} + F_{wx1} + F_{wx2} \quad (5)$$

Where, α_{11}, α_{12}, α_{13}, α_{21}, α_{22}, α_{23}, α_{31}, α_{32} are weighing factors used to speak to the significance of the important terms. Weighting elements are client characterized numbers extending from zero to one.

$F_{ws1}$and $F_{ws2}$are the weighted total of the pinnacle estimations of the RMS of the transmitted force and moments in the low and high recurrence band, individually.

$F_{wx1}$and F_{wx2} are the weighted total of the RMS of the transmitted force and moments barring the qualities compared to the pinnacle reaction as distinguished by F_{ws1} and F_{ws2}. F_{wi_rms}, M_{wxi_rms} and M_{wyi_rms} are the overall RMS value of the transmitted force and moments, along the vertical, roll and pitch axes.

Mathematical Model

A motor mounting framework comprising of three elastic-to-metal reinforced mounts is demonstrated by six

degrees of freedom and as shown in the schematic drawing of Figure 1. CG indicates the center of gravity, which is a reference point for static equilibrium. The baseline design of the mounts in this system is based on a commercial car engine subjected to an excitation force (f_e).

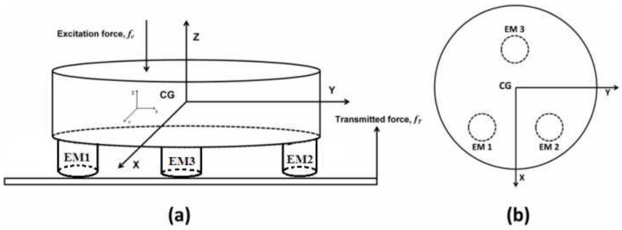

Figure 1: Engine Mounting System with Three Inclined Engine Mounts (EM): (a) Front View; (b) Top View [12]

The condition of movement for the motor mounting framework, which incorporates the frequency of subordinate properties, is given by condition

$$m_e\ddot{x} + [k(\omega)]x = f_e \quad (6)$$

Assume that the CG undergoes a small deflection, $[x]^T$ which represents the translational and rotational movement of the CG [k (ω)] is the stiffness matrix including the translational stiffness and rotational stiffness, and ½ f_e is the external excitation force.

To apply the force in a single direction, let $f_e = F_e e^{j\omega t}$and natural frequency be, $\omega_n = \sqrt{\frac{k}{m}}$. For the given design of mount, the excitation force and response occur at the same frequency. The dynamic stiffness k_D (ω), i.e. the ratio of the excitation force to the resulting deflection of the engine mounting system, is expressed as

$$k_D(\omega) = \frac{F_e}{x_o} = k(\omega)\left[\left\{1 - \frac{\omega^2}{\omega_n^2}\right\} + j\eta(\omega)\right] \quad (7)$$

Here, [k (ω)] is the frequency which depends on stiffness of the engine mount, and $\eta(\omega)$ is the frequency which depends on loss factor. Both of these quantities are the elemental properties of the individual engine mount.

For the frequency independent case, the force transmitted f_T to the foundation of the system is given by, $f_T = C\dot{x} + kx$ where c is damping coefficient and k stiffness of the engine mount. In case of hysteresis damping, the viscous damping coefficient is set as c = 0, and k is replaced with a complex and frequency dependent model, $F_T = k(\omega)[1 + j\eta(\omega)]$. The force transmitted to the foundation is,

$$f_T = k(\omega)\,[1 + j\eta(\omega)]x \quad (8)$$

Each individual engine mount in the global engine mounting system has a three dimensional complex stiffness matrix in the local coordinate system (xyz coordinates), which is given by the expression,

Stiffness in direction x, $k_x = \frac{2AG}{h}$ (9)

Stiffness in direction y, $k_y = \frac{2AG}{h}$ (10)

Stiffness in direction z, $k_z = \frac{2AG}{h}$ (E) (11)

Where, A is the average cross sectional area of engine mount, h is the height of engine mount, G is the shear modulus and E is the Young's modulus of the engine mount.

$$[k_D^{'}(\omega)] = \text{diagonal}[k_{Dx}(\omega) \quad k_{Dy}(\omega) \quad k_{Dz}(\omega)] \tag{12}$$

$$[k_T^{'}(\omega)] = \text{diagonal}\ [k_{Tx}(\omega) \quad k_{Ty}(\omega) \quad k_{Tz}(\omega)] \tag{13}$$

These stiffness matrixes must be changed from local coordinate system to global coordinate system by using Euler transformation matrix [A] which is given by,

$$[A] = \begin{bmatrix} \cos\theta_x cos\theta_y & -\sin\theta_x cos\theta_z + \cos\theta_x sin\theta_y \sin\theta_z & \sin\theta_x sin\theta_z + \cos\theta_x sin\theta_y \cos\theta_z \\ \sin\theta_x cos\theta_y & \cos\theta_x cos\theta_z + \sin\theta_x \sin\theta_y \sin\theta_z & -\cos\theta_x sin\theta_z + \sin\theta_x sin\theta_y \cos\theta_z \\ -sin\theta_y & \cos\theta_y \sin\theta_z & \cos\theta_x \cos\theta_z \end{bmatrix} \tag{14}$$

Where, θx, θy and θz are the orientation angles of the mount measured from $_{x,\ y}$ and $_{z}$ axis. The transformation from the local stiffness matrix to the global stiffness matrix is performed using the following expressions:

$$[k_{Di}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})] = [A]\ [k_{Di}^{'}]\ [A]^{-1} = \begin{bmatrix} k_{Dxxi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Dxyi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Dxzi}(\omega, \theta_x, \theta_y, \theta_z) \\ k_{Dyxi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Dyyi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Dyzi}(\omega, \theta_x, \theta_y, \theta_z) \\ k_{Dzxi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Dzyi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Dzzi}(\omega, \theta_x, \theta_y, \theta_z) \end{bmatrix} \tag{15}$$

$$[k_{Ti}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})] = [A]\ [k_{Ti}^{'}]\ [A]^{-1} = \begin{bmatrix} k_{Txxi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Txyi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Txzi}(\omega, \theta_x, \theta_y, \theta_z) \\ k_{Tyxi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Tyyi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Tyzi}(\omega, \theta_x, \theta_y, \theta_z) \\ k_{Tzxi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Tzyi}(\omega, \theta_x, \theta_y, \theta_z) & k_{Tzzi}(\omega, \theta_x, \theta_y, \theta_z) \end{bmatrix} \tag{16}$$

Where, The subscripts for the matrix element, xx, xy, xz,…zz, show the element properties in a particular direction. However, only the principal directions (X, Y and Z) are considered for both the driving point and the transfer stiffness. i =1, 2 ….is the number of the engine mount.

The summation of, dynamic properties of each mount is then carried out to obtain the global properties of the mounting system by using position matrices of each individual mount $[r_i]$:

$$[r_i] = \begin{bmatrix} 0 & z_i & -y_i \\ -z_i & 0 & x_i \\ y_i & -x_i & 0 \end{bmatrix} \tag{17}$$

Finally, the total dynamic stiffness and transfer stiffness in the global coordinate system are obtained by summation of the individual mount properties and individual position matrices.

$$\left[k_D(\omega)_{(global)}\right] = \begin{bmatrix} \sum_{i=1}^{n}[k_{Di}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})] & -\sum_{i=1}^{n}[k_{Di}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})r_i] \\ -\sum_{i=1}^{n} r_i^T [k_{Di}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})] & -\sum_{i=1}^{n} r_i^T [k_{Di}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})] \end{bmatrix} \tag{18}$$

$$\left[k_T(\omega)_{(global)}\right] = \begin{bmatrix} \sum_{i=1}^{n}[k_{Ti}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})] & -\sum_{i=1}^{n}[k_{Ti}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})r_i] \\ -\sum_{i=1}^{n} r_i^T [k_{Ti}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})] & -\sum_{i=1}^{n} r_i^T [k_{Ti}(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi})] \end{bmatrix} \tag{19}$$

The force transmissibility ratio of the system, T of the system in a single direction is the ratio of the transmitted force to the excitation force of the system and is obtained as,

$$T = \frac{F_T}{F_e} = \frac{k(\omega)\ [1 + j\eta(\omega)]}{k(\omega)\left[\left\{1 - \frac{\omega^2}{\omega_n^2}\right\} + j\eta(\omega)\right]} \tag{20}$$

The transmissibility of force in each direction could be found by using the equation,

Transmissibility of force in direction x,

$$T_x\,(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi}, x_i, y_i, z_i) = \frac{k_T(\omega)_{(global)}[1,1]}{k_D(\omega)_{(global)}[1,1]} \tag{21}$$

Transmissibility of force in direction y,

$$T_y\,(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi}, x_i, y_i, z_i) = \frac{k_T(\omega)_{(global)}[2,2]}{k_D(\omega)_{(global)}[2,2]} \tag{22}$$

Transmissibility of force in direction z,

$$T_z\,(\omega, \theta_{xi}, \theta_{yi}, \theta_{zi}, x_i, y_i, z_i) = \frac{k_T(\omega)_{(global)}[3,3]}{k_D(\omega)_{(global)}[3,3]} \tag{23}$$

3. OPTIMIZATION METHOD

The very purpose of this work is to optimize the model which is taking continuous cyclic loading (vibration) such that the stress induced on the mount is minimum and life of the product become higher. The optimization and analysis carried out in Ansys R17.0. The engine selected for doing the optimization and analysis has 170 kg weight, torque of the engine is 10^2 Nm, maximum idling speed is 500 rpm and maximum speed is 9000 rpm [4]. Existing model of passive engine mount is made of silicon elastomer material with following properties as shown in Table 1.

Table 1: Material Properties of Silicon Elastomer

Sl. No.	Properties	Notation	Values
1.	Young's modulus	E	50 vMPa
2.	Poison ratio	μ	0.5
3.	Density	ρ	2300 kg/m^3
4.	Bulk modulus	K	2000 MPa
5.	Compressive strength	σ_c	20 MPa
6.	Shore hardness value	H	65 HA
7.	Shear modulus	G	16.6MPa

Finite Element Model

The mount used here is common in case of any passenger cars. The model (both existing and optimized) is built by using brick 8 node 185 and also tetrahedral element.

The one end of the mount is connected to engine base and other end is connected to engine mount bracket or directly fixed on the steel hood of the wheel through the hole provided in the mount by using steel screw rod. The boundary condition is applied by constraining all the surface of the engine mount.

Figure 2 shows the commercially available engine mount, initially it is modeled by using Catia and then it is converted into Ansys file by using igs extension. The igs file is then directly opened in Ansys and number of division for each line is given and it is map meshed as shown in Figure 3.

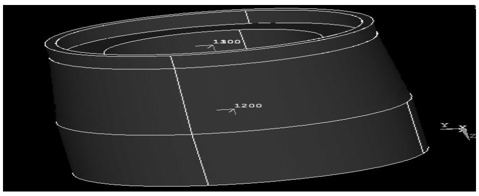

Figure 2: Model of Existing Rubber Engine Mount

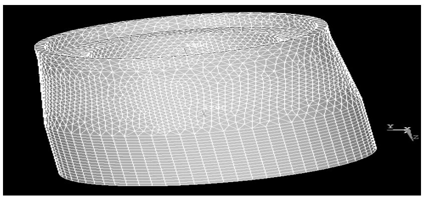

Figure 3: Existing Rubber Engine Mount FEA Model with Mapped Meshing

3.2. Shape and Parameter Optimization

The main parameter considered for optimization is stiffness which is determined by the static structural analysis and the stiffness obtained from the geometry of the mount. The stiffness is considered as main parameter as it helps in reducing Noise, Vibration and Harshness (NVH) characteristics of the engine.

Figure 4 shows the commercially available engine mount, initially it is modeled by using Catia and then it is converted into Ansys file by using.igs extension. Then this.igs file is fed into Ansys workbench for direct optimization by giving necessary boundary condition then the optimized file is saved in the computer. Then it is opened in Ansys and number of division for each line is given and it is map meshed as shown in Figure 5.

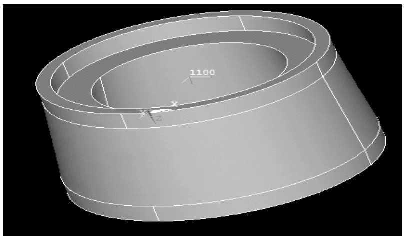

Figure 4: Model of Optimized Rubber Engine Mount

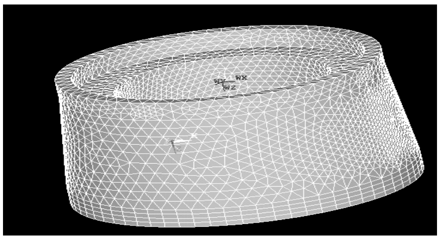

Figure 5: Optimized Rubber Engine Mount FEA Model with Mapped Meshing

4. STRESS AND STRAIN ANALYSIS

Detailed stress and strain analysis of rubber part is done in this paper. For analysis purpose each of the rubber mount is subjected to a load of 555.9 N and the Von Mises stress distribution is obtained as shown in Figure 6 and Figure 7 for existing model and the optimized model.

4.1. For Existing Model

The stress and strain analysis is carried out on the existing model. The response of the structure and notes are assumed to vary very slowly with respect to time. The maximum strain undergone by the mount is 6.4294. The maximum value of stress (Von Mises stress) undergone by the mount is 484 MPa in the top region (as shown in the Figure 4). Hence the stress acting on the top portion of the mount is very high.

4.2. For Optimized Model

The strain acting on the optimized model is 5.6524. The maximum value of stress (Von Mises stress) undergone by the mount is 330.104 MPa hence the stress and strain analysis result for the optimized model is much better than existing model.

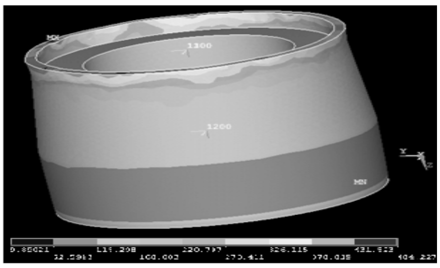

Figure 6: Von Mises Stress Distribution in Existing Model

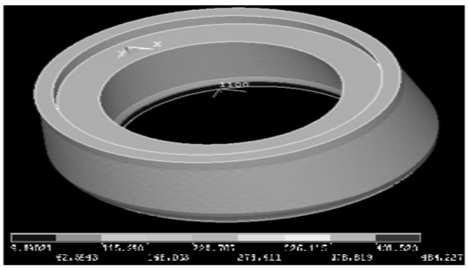

Figure7: Von Mises Stress Distribution in Optimized Model

5. RESULTS AND DISCUSSIONS

The existing model is optimized for shape and also for the stiffness parameter. The optimized shape of the engine mount is as shown below in Figure 8. This result is obtained from the software by the method of direct optimization of the nonlinear elements. In the shape it could be observed that the shape has been modified to an optimum value to get the best result when the cyclic load is being applied on it [6, 12]. The result obtained for stiffness of the engine mount is given in Table 2.

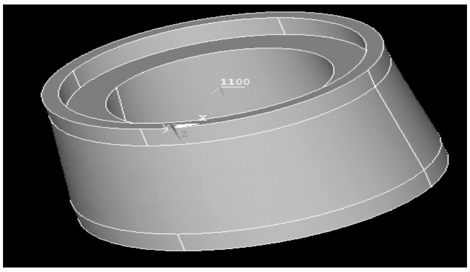

Figure 8: Optimized Shape of Mount

Table 2: The Value of the Stiffness of Existing and Optimized Model

		Existing Model	Optimize Model
Stiffness	k_x	1183.58	563.69
	k_y	1185.98	556.34
	k_z	6317.62	9765.97

Stress and stiffness in the direction x & y decreased by optimized model. It will transmit the least amount of vibration from the engine block to the passenger compartment. And also the stiffness in the direction is increased which means deformation in z direction is very less [6]-[12]-[13].

From the Table 2, it could be observed that the value of the stiffness in direction x and the stiffness in the direction y remains to be same as predicted in the mathematical model and stiffness on the direction z depends mainly on the design of the mount as well as the strength of the material [14].

In Table 2, original parameter of the existing model for optimization and the optimize parameter is given. Since, the space is limited for the engine bay, the engine mount location is also constrained. The constrained include passive engine mount stiffness, damping coefficient and mounting location. Hence, in this paper the stiffness is optimized [6].

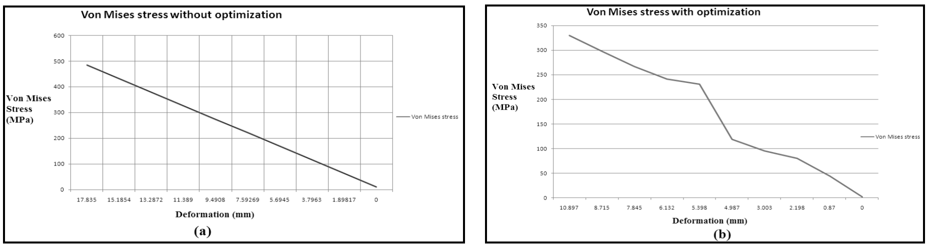

Figure 9: Von Mises Stress Versus Displacement Curve
a) for Existing Model b) for Optimized Model

From the above graph (Figure 9) it could be observed that the Von Mises stress induced in the optimized model is less than that of the stress induced in the existing model. And also it could be observed that the elastic deformation for the optimized model is also less than that of the existing model. These effects could be observed, because of shape

optimization. Since the shape is in optimum condition the stress and strain induced in them would be reduced.

A detailed stress-strain analysis is rubber part is analyzed in this work. From the above curve it could be observed that, the maximum Von Mises stress for the existing model is more than the stress in the optimized model. The Von Mises stress for the metal is given by the equation

$$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 = 2\sigma_k^2 \tag{24}$$

Yielding in metal occurs due to shear strain energy. This equation is not suitable for polymer such as rubber because hydrostatic stress of rubber is not taken into account. So, the Von Mises stress for polymer is modified by considering the hydro statistic. The equation is given by

$$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 + 2(\sigma_c - \sigma_T)(\sigma_1 + \sigma_2 + \sigma_3) = 2\sigma_c\sigma_T \tag{25}$$

Were the hydro statistic stress is given by$(\sigma_1 + \sigma_2 + \sigma_3)$; σ_c and σ_T are absolute value of compressive and tensile yield stress respectively, but there is no general yield criteria for polymer rubber [6]-[10]-[11]-[12]-[13].

The engine and mount displacement include both static and dynamic motion for center of engine mass and passive mount is constrained owing limit to engine bay [6].

The maximum load bearing capacity of the silicon rubber material when it is subjected to cyclic loading is 800MPa [12]. The optimized shape obtained is subjected to a cyclic load well within that limit (i.e. 330Mpa) so the design is also safe and also it is very less than the non-optimized shape.

6. CONCLUSIONS

The optimization and analysis of the engine mount is successfully completed. By this method of optimization the life of the passive rubber mount is also increased since the top and bottom portion undergoes less deformation and less vibration transmitted from the engine to the passenger compartment since the stiffness is optimized. In future similar method could be used to optimize some other engine mount.

- In the present case the effect (Von Mises) stress is regarded as a measure of yielding rubber spring. However, its effectiveness needs to be verified by future experiments.
- The static characteristics mount depends on the load.
- Experimental work needs to be conducted to validate the optimized model and simulate the results for engine mounting system.
- In order to evaluate further performance of the passive engine mount a full scale or a scale down vehicle model is recommended.

REFERENCES

1. *Yunhe Yu, Nagi G. Naganathan, Rao V. Dukkipati,A literature review of automotive vehicle engine mounting,Mechanism and Machine Theory,Vol. 36, n. 1, pp. 123-142, 2001. https://doi.org/10.1016/S0094-114X(00)00023-9*
2. *Gang Sheng, vehicle noise, vibration and sound quality (1st edition, SAE International, 2012).*
3. *R. P. Sharma, a course in automobile engineering (1st edition, Dhanpath Rai publication, 1998).*

4. *Monali Deshmukh, Prof. K R Sontakke, Analysis and Optimization of Engine Mounting Bracket,International Journal of Scientific Engineering and Research (IJSER), Vol. 3, n. 5, pp.131 – 136, 2015.*
5. *Kamal Jahani, Masoud Dehnad, Identifying the frequency dependent material property of a hydraulic engine mount,Journal of Mechanical Science and Technology, Vol. 28, n. 6, pp. 2041-2047, 2014.* http://link.springer.com/article/10.1007/s12206-014-0111-9
6. *B.Ramesh et al.,, Aero-Structural Optimization of Aircraft Wing by Finite Element Method, International Journal of Mechanical and Production Engineering Research and Development (IJMPERD), Volume 7, Issue 6, November - December 2013, pp. 409-416*
7. *Wang Ruiping, A study of vibration isolation of engine mount system, Master of Applied Science thesis, Concordia University Montreal, Quebec, Canada, 2005.*
8. *Baocheng Zhang, Haifei Zhan,YuantongGu, A general approach to tune the vibration properties of the mounting system in the high speed and heavy duty engine,Journal of Vibration and Control,Vol. 22, n. 1, pp. 247 – 257, 2016. http://journals.sagepub.com/doi/abs/10.1177/1077546314528963*
9. *Wen-Bin Shangguan, Da-Ming Chen, A method for calculating the displacements of a powertrain and mounts in a powertrain mounting system including torque struts under quasi-static loads,Journal of automobile engineering,Vol. 226, n. D, pp. 634 – 647, 2011. http://journals.sagepub.com/doi/abs/10.1177/0954407011424816*
10. *G. Kim, R. Singh, A study of passive and adaptive hydraulic engine mounts systems with emphasis on non-linear characteristics,Journal of Sound and Vibration, Vol. 179, n. 3, pp. 427 – 453, 1995. https://doi.org/10.1006/jsvi.1995.0028*
11. *L-R Wang, Z-H Lu,I Hagiwara, Finite element simulations of the static characteristics of a vehicle rubber mount, Journal of automobile engineering, Vol. 216, n. D, pp. 965 – 973, 2002. http://journals.sagepub.com/doi/abs/10.1243/095440702762508218*
12. *Iulian Grindeanu, Nam H. Kim, Kyung K. Choi,JiunS. Chen, CAD-Based Shape Optimization Using a Meshfree Method, CONCURRENT ENGINEERING: Research and Applications, Vol. 10, n. 1, pp. 55 – 66, 2002. http://journals.sagepub.com/doi/abs/10.1177/1063293X0201001056*
13. *Lu-EanOoi, Zaidi MohdRipin, Optimization of an engine mounting system with consideration of frequencydependent stiffness and loss factor,Journal of Vibration and Control, Vol. 22, n. 10, pp. 2406–2419, 2016. http://journals.sagepub.com/doi/abs/10.1177/1077546314547532.*
14. *J J Kim, H Y Kim, Shape design ofanengine mount by a method of parametric shape optimization, Proc InstnMechEngrs, Vol. 211, n. D, pp. 155–159, 1997. https://doi.org/10.1016/S0045-7949(95)00118-2*
15. *A. S. Sathawane, A. V. Patil, Analytical Study of Engine Mount to Suit the Damping Requirement of the Engine, International Journal of Latest Trends in Engineering and Technology, Vol. 3, n. 3, pp. 7-10, 2014.*